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A TRANSIENT METHOD OF NEGLIGIBLE INTERNAL THERMAL
RESISTANCE FOR DETERMINING THERMAL CONTACT
CONDUCTANCE IN A VACUUM

BY

RICHARD CHUKA OBOKA, 1941-

A

THESIS

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Approved by

Orlando Mr. May

(advisor)

D. Gyong

C. Y. Ho

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ABSTRACT

A transient method for experimentally determining the interface conductance between metallic surfaces in contact is developed. The method applies the theory of negligible internal thermal resistance of one of the materials which form the interface. The method is applicable when the interface conductance is very low, a situation which exists when low conductance interstitial materials are used between contacting surfaces for purposes of thermal isolation.

In this investigation, stainless steel wire screens of 10 and 100 mesh were used as interstitial materials between aluminum surfaces in a vacuum environment of 10^{-5} to 10^{-6} Torr. The interface pressure ranged from 40 to 200 psi. The roughness of the contacting surfaces ranged from 15 to 25 micro inches, root mean square.

The experimental results show that the average thermal conductance obtained by the transient method was from 96.04 to 98.5 % of the steady state thermal conductance when the 10 mesh wire screen was used as the interstitial material and was from 79.28 to 91.13 % of the steady state thermal conductance when the 100 mesh wire screen was used. The results thus show that the method of negligible internal thermal resistance is quite reliable for very low interface conductances.

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I. INTRODUCTION

The existence of an additional thermal resistance at the interface formed by two similar or dissimilar metallic surfaces in contact has been established for sometime. This additional resistance is known as thermal contact resistance.

A knowledge of the effects of thermal contact resistance is very important in many engineering applications. The thermal design of systems such as spacecrafts, nuclear reactors and cryogenic storage tanks requires a good understanding of thermal contact resistance in order to avoid an unexpected failure of the system due to excessive thermal stresses or temperatures. In space technology, the reliability of electronic, electrical and mechanical components depends on accurate thermal design in which thermal contact analysis plays an important role.

The fact that thermal contact resistance exists in many engineering systems leads to the question of its controllability, that is to say, what is the possibility of increasing or decreasing the thermal resistance between surfaces in contact ?. This is obviously an important question since in nuclear reactors, for example, high heat flux is essential for good operation and thus it would be desirable to decrease or to eliminate any existing thermal contact resistances in such applications. However, in spacecraft technology where the thermal isolation of such components as reflective shields and antenna struts is important, one

would like to increase any existing thermal contact resistances in order to decrease the heat flux, thereby providing some insulating effects. Numerous investigations have been conducted to determine whether thermal contact resistance can be successfully increased or decreased by introducing materials between the contacting surfaces of various similar and dissimilar metals, and the results have been quite promising.

Almost all previous investigators have determined thermal contact conductance by means of steady state experiments, using the one-dimensional, steady state heat equation for their calculations. Steady state experiments usually involve very long time intervals between readings while transient experiments involve very short time intervals. The object of this investigation is to determine the feasibility of using a transient method of negligible internal thermal resistance to measure the thermal contact resistance of members in a vacuum environment. Thermal contact resistance cannot be accurately predicted due to the many parameters which are involved; thus, a great deal of experimentation is necessary. The proposed transient method has the advantage of providing reliable estimates of thermal contact resistance in a very short time. It is particularly applicable in thermal isolation problems which usually involve very high contact resistances.

II. LITERATURE SURVEY

The problem of thermal contact resistance has received considerable attention within the last few decades. The problem has been attacked both experimentally and analytically and much information has been published thus far. However, the phenomenon of thermal contact resistance still requires further exploration due to the many parameters which are involved.

The heat transfer through an interface formed by two materials in contact is composed of three modes: i) solid conduction through the actual area of metallic contact, ii) conduction through the interstitial filler, and iii) thermal radiation. Barzelay, Tong and Holloway (1)* reported that none of the three modes seemed to play a dominant role in contributing to the total heat transfer across a metallic interface. Their investigation was carried out under the presence of atmospheric air. Clausing and Chao (2) investigated the mechanism of heat transfer at metallic interfaces in a vacuum environment and reported that solid conduction accounted for approximately the total heat transfer at the interface. In a vacuum environment, only solid conduction and thermal radiation contribute to the total heat transfer across an interface. At very low contact pressures, thermal radiation might be important; but at high pressures, the

* Numbers in parentheses refer to listings under References.

actual contact area increases thus making solid conduction a dominant mode of heat transfer.

The thermal contact resistance at the interface depends on such factors as i) the roughness of the mating surfaces, ii) the mean interface temperature, iii) the interface pressure, iv) the material properties of the mating surfaces, v) the nature of the interstitial filler, and vi) the mechanical and thermal boundary conditions of the solid. Barzelay, Tong and Holloway (3) investigated the effects of surface roughness, interface pressure and mean interface temperature on the thermal contact resistance of aluminum-aluminum and stainless steel- stainless steel joints. They reported that the smoother the surfaces in contact, the lower the contact resistance and that contact resistance decreases as the mean interface temperature and the interface pressure increase. Other investigators such as Atkins and Fried, Fry, Clausing and Chao and Stubstad reported similar trends in vacuum environment.

Rogers (4) investigated the contact resistance at the interface of dissimilar metals in air and in vacuum. He used aluminum-steel interfaces and observed that the contact resistance at the interface depended on whether the heat flow direction was from steel to aluminum or vice versa. He also noted that the magnitude of the thermal contact resistance was much higher in vacuum than in air irrespective of the heat flow direction. The tests were performed

at a constant interface pressure of 122 psi. Petri (5) conducted similar tests using molybdenum- aluminum members. The heat flow direction was from molybdenum to aluminum. For pressures up to 160 psi, he observed only a very slight difference between the conductances in air and the conductances in vacuum. For high pressures i.e., pressures above 500 psi, he observed the same values in air as in vacuum.

Several investigators have surveyed the effects of interstitial materials on thermal contact resistance. Barzelay, Tong and Holloway reported that a 0.001 inch brass foil placed between the rough stainless steel interfaces increased the thermal conductance at high pressures while the same foil between aluminum interfaces decreased the thermal conductance. This resulted from the fact that the brass foil was softer than the stainless steel members but was harder than the aluminum members. They also noted that a 0.01 inch asbestos sheet lowered the conductance between stainless steel surfaces by as much as 80 % at all pressure levels. Tests were made in atmospheric air. Koh and John (6) conducted tests to find the effects of low and high conductivity metallic foils on thermal contact resistance in air. They used copper, aluminum, lead and indium foils between mild steel-mild steel interface. Their results indicated that the softness of the foil material rather than the thermal conductivity was the important factor in reducing the thermal contact resistance. They

also observed that there existed an optimum foil thickness for use between surfaces of a given roughness so as to provide a maximum reduction in contact resistance.

Smuda and Gyorog (7) conducted investigations with several low conductance interstitial materials. They conducted their experiments in a vacuum using aluminum-aluminum and stainless steel-stainless steel interfaces. The interstitial materials used included silica paper, asbestos board, mica, carbon paper, laminate T-30 LR, WRP-X-AQ felt and stainless steel wire screens. All the interstitial materials tested exhibited good insulating characteristics with carbon paper acting as the best thermal barrier. The results indicated that the choice of a particular low conductance interstitial material would depend on the expected application since some of these materials have limiting mechanical strength. Because of their high mechanical strength, stainless steel wire screens were suggested for high load applications. Further investigation was made by Gyorog (8) with wire screens as interstitial materials. He observed that the resistance offered by wire screens could be accurately predicted from empirically derived dimensionless parameters and that the contact resistance depended on the mesh size, the diameter and the material properties of the wire screen and the interface pressure.

Schauer and Giedt (9) described an experimental method to determine the interface conductance between two plates of dissimilar metals during transient heating. This method involved heating one of the plates with a capacitor-bank discharge and recording the contacting-surface temperatures of the two plates as they came to equilibrium. They reported that for metallic contacts (aluminum-stainless steel) the conductance increased about 200 % with time and that the reverse behavior occurred with the metal-ceramic contacts. The interface conductance approached an asymptotic value (steady state value) in each case within 180 milliseconds.

III. THEORETICAL TREATMENT

The theoretical model is schematically shown in Figure 1. It consists of symmetrically constructed heat sinks with the specimen between them. The sinks and the specimen are made of the same material (aluminum 6061-T6). The heat sinks are assumed to be at the same temperature, T_s . The specimen is so small that the contact pressure is assumed to be the same at the two interfaces. All the interfaces are assumed to have the same thermal and mechanical boundary conditions and thus the same thermal contact conductance.

In a vacuum environment, the energy balance for the specimen in the interval of time dt is given by

$$-cVdt = hA (T - T_s)dt \quad (1)$$

where

c = specific heat of specimen, Btu / cu.ft $^{\circ}\text{F}$

V = volume of specimen, cu. ft

A = area of specimen exposed to heat transfer, sq. ft

T = average temperature of specimen, $^{\circ}\text{F}$

T_s = temperature of sink, $^{\circ}\text{F}$

h = average thermal contact conductance, Btu / hrsq.ft $^{\circ}\text{F}$

dt = change in time, hr

dT = change in temperature of specimen in time dt , $^{\circ}\text{F}$.

In writing equation (1), it was assumed that (i) the in-

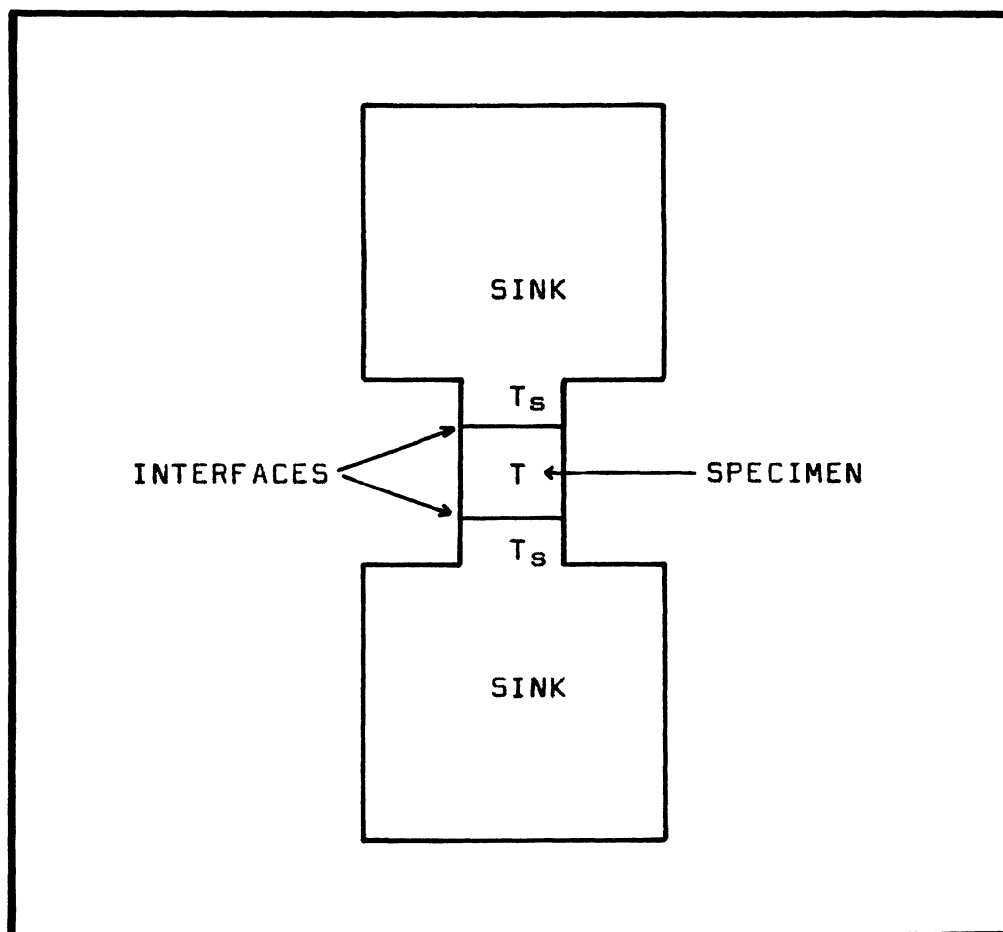


Fig. 1 Theoretical Model

ternal conductive resistance of the specimen is uniform at all times, (ii) the external thermal resistance between the heat transferring surfaces of the specimen and of the surrounding medium is very large compared to the internal thermal resistance of the specimen, (iii) the radiation heat loss from the specimen to the surroundings is negligible. Furthermore the thermal contact conductance was assumed to be constant.

For this problem, $T_s = T_s(t)$. Hence equation (1) becomes

$$dT / dt + mT(t) = mT_s(t) \quad (2)$$

where

$$m = hA / cV. \quad (3)$$

This is a first order linear non-homogeneous differential equation with constant coefficients. Integrating equation (2) yields

$$T(t) = B_1 e^{-mt} + m e^{-mt} \int e^{mt} T_s(t) dt \quad (4)$$

where B_1 can be determined by the initial condition, viz: at time, $t = 0$, $T(0) = T_0$, and $T_s(t)$ is determined experimentally.

From the experiment, the variation of the sink temperature with time was found to be of the form

$$T_s(t) = C_1 t^2 + C_2 t + C_3 \quad (5)$$

where C_i , $i = 1, 2, 3$, are known constants. Substituting equation (5) into equation (4) yields on integrating

$$T(t) = B_1 e^{-mt} + C_1 t^2 + (C_2 - 2C_1 / m)t + 2C_1 / m^2 - C_2 / m + C_3. \quad (6)$$

Applying the initial condition that $T(0)=T_0$ at time, $t=0$, the constant, B_1 , is found to be

$$B_1 = T_0 - (2C_1 / m^2 - C_2 / m + C_3). \quad (7)$$

Thus the final expression for the temperature of the specimen can be written as ,

$$T(t) = (T_0 - (2C_1 / m^2 - C_2 / m + C_3))e^{-mt} + C_1 t^2 + (C_2 - 2C_1 / m)t + 2C_1 / m^2 - C_2 / m + C_3. \quad (8)$$

In equation (8) $T(t)$ can be obtained by experimental measurements, thus leaving m as the only unknown. An iterative scheme is employed to solve equation (8) for m ; consequently, the interface thermal conductance h is obtained from equation (3). The iterative scheme used is described in Appendix A2.

IV. DESCRIPTION OF TEST APPARATUS

The apparatus consisted mainly of a cylindrical test specimen, a cartridge heater, two 2-step cylindrical heat sinks and a lever type loading system. A schematic representation of the test fixture is shown in Figure 2.

(i) Test Specimens

The test specimens used were 6061-T6 aluminum cylinders, 1 inch in diameter and 0.25 inch in length. The specimens were placed between two similarly constructed heat sinks, thus providing two interfaces. Two interfaces were used in order to provide more surface area of the specimen for heat transfer and thus to decrease the significant length of the specimen. The significant length is the ratio of the volume of the specimen to the area of the specimen exposed to heat transfer. All the surfaces which made up the interfaces were machined carefully on a lathe at the same feed rate so that the surfaces would approximate the same degree of roughness. The surface roughness of all the contacting surfaces ranged from about 15 to 25 micro-inches, root mean square, as measured by a Bendix Micrometrical Profilometer. A cartridge heater was installed diametrically inside the specimen as shown in Figure 3. The thermocouple holes, 0.0625 inch in diameter and 0.1875 inch in depth, were drilled at distances of 0.0625 inch from the flat surfaces of the specimen. The holes were 0.125 inch apart and were drilled parallel to

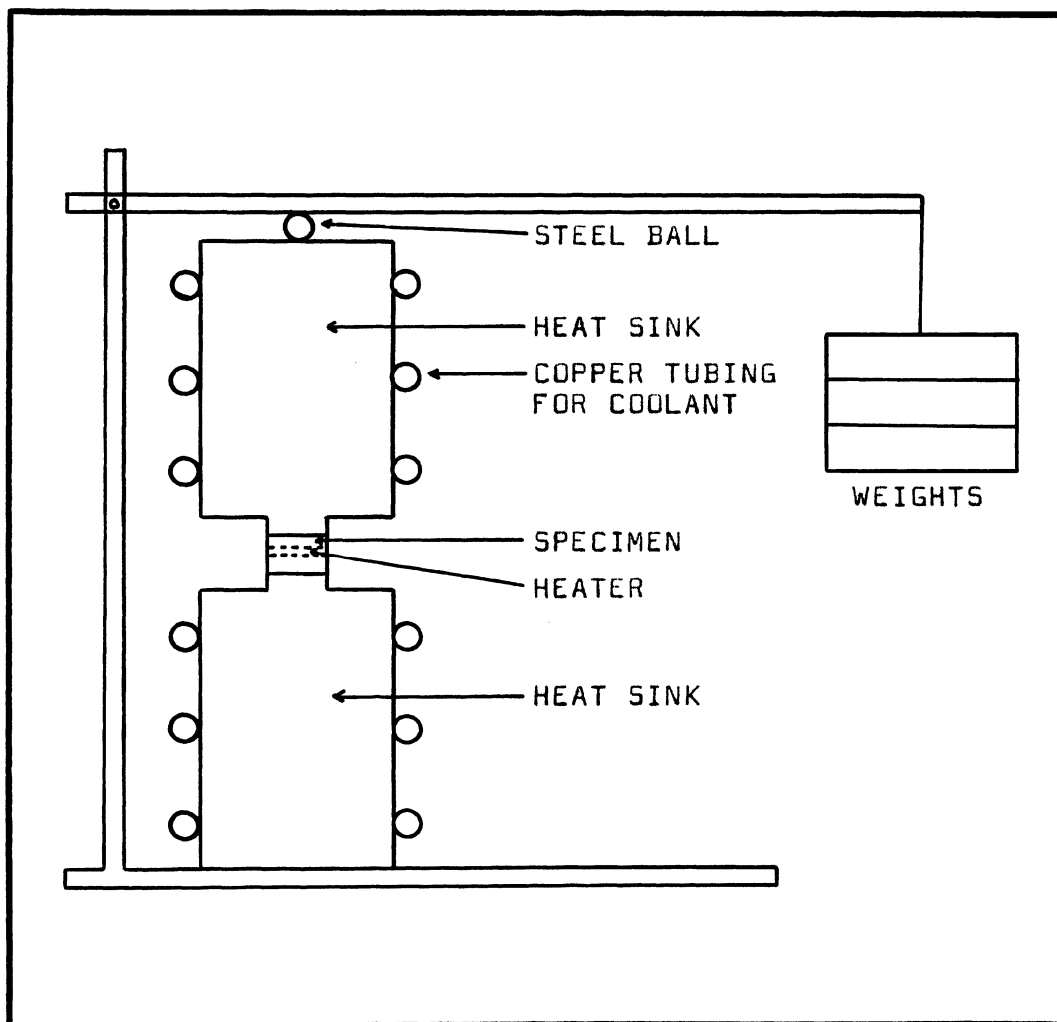


Fig. 2 Test Fixture

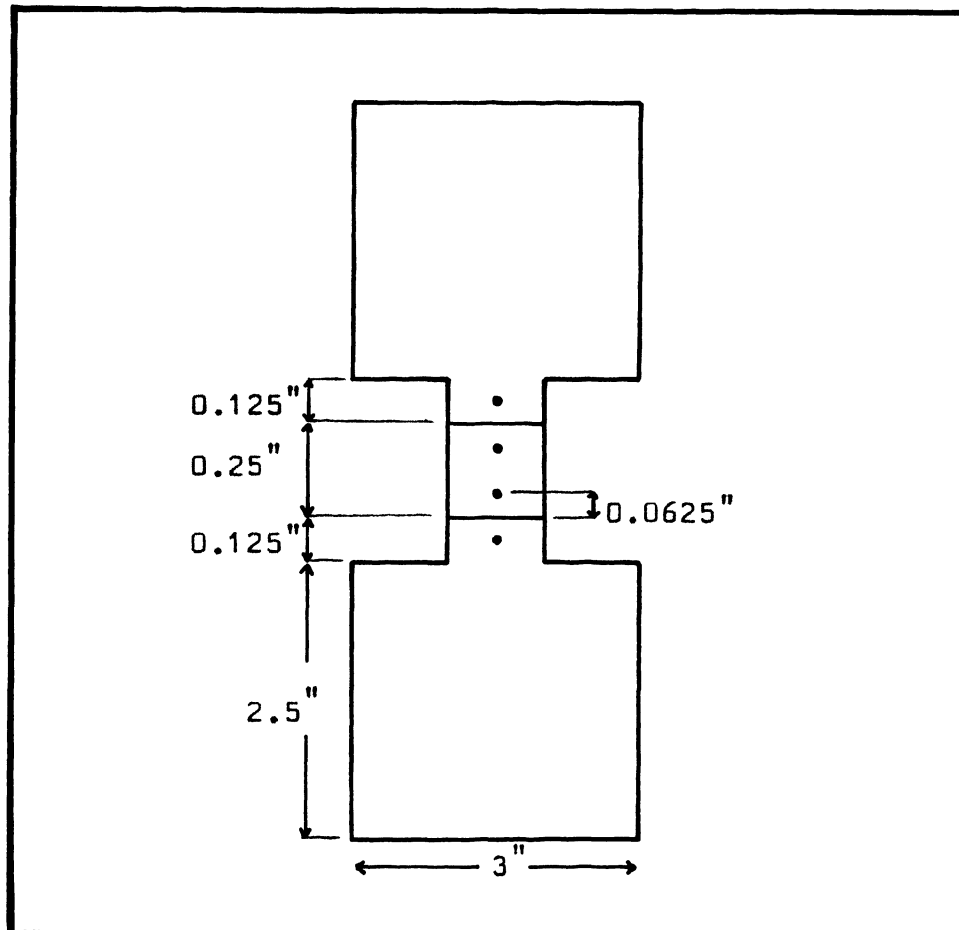


Fig. 3 Significant Dimensions and Thermocouple Positions

the flat surfaces and perpendicular to the axis of the cylindrical specimen.

(ii) Heat Source

The heat source used was a 15-watt cartridge heater, 0.125 inch in diameter and 1 inch in length, supplied by Hotwatt, Inc.. The heater was installed diametrically inside the specimen. A Heathkit Regulated Power Supply was used to supply electrical power to the heater. The voltage and current registered on the Heathkit Regulated Power Supply were used to calculate the power input to the heater.

(iii) Heat Sinks

Two 6061-T6 aluminum 2-step cylinders with major dimensions of 3 inches in diameter and 2.5 inches in length and minor dimensions of 1 inch in diameter and 0.125 inch in length were used as the heat sinks. The 2-step cylinders were identically wrapped with high conductivity copper tubing through which cooling water was circulated. The heat sinks were also utilized to furnish part of the contact surfaces. The use of the 2-step cylinders as the heat sinks was to aid in insuring one-dimensional heat flow through the contact surfaces. A thermocouple was installed in each of the 2-step cylinders to measure the sink temperature.

(iv) Temperature Measuring Device

Four iron-constantan thermocouples with 30 gage wires

were used to measure the temperatures. The thermocouples were installed at distances of 0.0625 inch on either side of the interfaces. They were held in place in 0.0625 inch diameter and 0.1875 inch deep holes by means of epoxy. All thermocouple junctions were made with DYNATECH Thermocouple Welder Model 116. An ice bath at 32⁰F was used as the reference temperature. The thermocouples were connected through a selector switch to chart recorders which measured the transient temperature in terms of millivolts.

(v) Loading System

A lever type loading mechanism was designed to provide the interface pressure. The loading system comprised primarily of a fulcrum, a loading pin which insured an axial concentrated load and a loading arm with weight attached to its end. With this loading system it was possible to obtain pressures of 40 to 200 psi at the interface. All pressures were calculated.

(vi) Radiation Shield

The isolation of the test specimen from radiation was accomplished by wrapping the specimen with Milar sheet. For the temperature range which prevailed during the experiment (about 60⁰F to 150⁰F) heat losses by radiation were very small compared to the heat flow through the interfaces (2.0 %). Experimental determination of the radiation heat losses is described in Appendix 1.

(vii) Wire Screens

Stainless steel wire screen with 10 and 100 mesh per linear inch were used at the interfaces. The 10 mesh screen had a diameter of 0.025 inch and 56.3 % open area and the 100 mesh screen had a wire diameter of 0.004 inch and 36 % open area.

(viii) Vacuum System

The entire test assembly was installed in the bell jar of a Varian 1G10 vacuum system equipped with a diffusion pump capable of maintaining pressures as low as 10^{-10} Torr.

V. TEST PROCEDURE

Two types of tests were performed. The first was conducted with 100 mesh stainless steel wire screens at the two interfaces formed between the specimen and the two heat sinks and the second was conducted with the 10 mesh stainless steel wire screens at the interfaces. Special care was taken to see that the wire screens and the contacting surfaces were properly aligned and that the load pin was well aligned with the axis of the specimen to insure axial loads.

For each test, runs were made for five different pressures, viz: 40, 80, 120, 160, and 200 psi. Since the loading system was completely enclosed in the vacuum chamber, it was necessary to devacuum the chamber after each run in order to change the interface pressure. Extreme care was therefore exercised in order to keep the interfaces undisturbed between runs. For each run, a vacuum of 10^{-5} to 10^{-6} Torr was created in the chamber. The specimen was then heated by a constant power supply. In order to compare the transient method with the steady state method, it was necessary to obtain steady state readings. A period of about three hours was allowed for steady state conditions to be reached. The voltage and current registered on the power supply unit were recorded at steady state and the temperature on either side of the interfaces was monitored. The

steady state conductance h_s was then evaluated as

$$h_s = Q / (A_s (\Delta T))$$

where Q is the heat supplied by the heater, $A_s = 2A$ where A is the cross sectional area of the specimen and ΔT is the temperature change at the interface.

After the steady state reading had been taken, the heater was turned off and the specimen was allowed to cool. The temperatures of the specimen and of the heat sink were continuously monitored on the chart recorders as functions of time as the specimen cooled. The temperature-time history of the heat sink was used in equation (4) to obtain the constants C_1 , C_2 , and C_3 and the temperature-time history of the specimen was used in equation (7) to evaluate m and thus h , the thermal contact conductance. A typical temperature-time history of the specimen and of the sink is shown in Figure 4. The same procedure was followed for all runs.

Throughout the tests, a constant flow rate of cooling water was maintained through the copper tubing around the heat sinks in order to keep the two sinks at the same temperature. The maximum temperature difference between the two heat sinks was about 1 °F during the entire course of the tests and the temperature of the specimen remained uniform to within 1 °F throughout. This represented less than 1 % variation of the temperature.

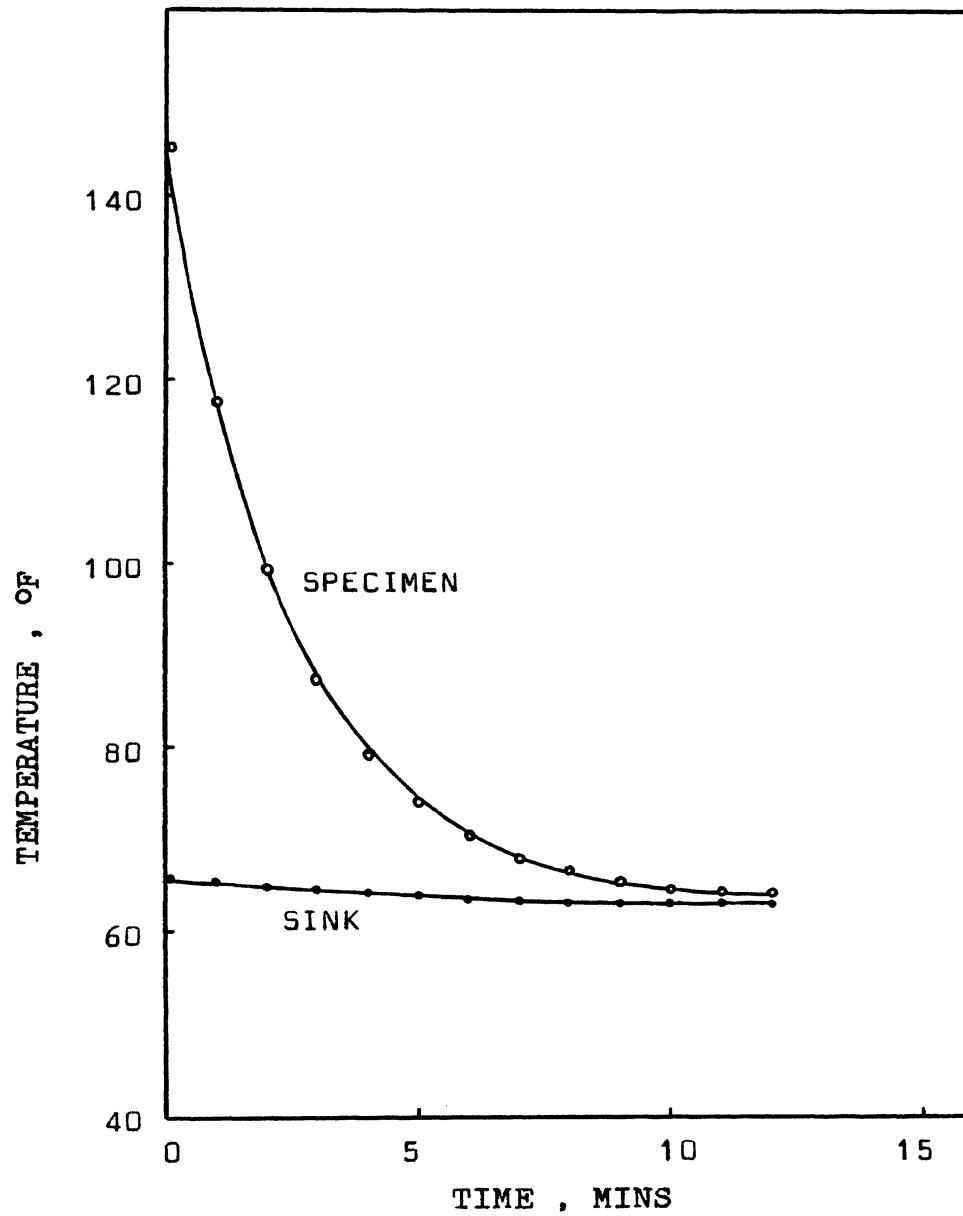


Fig. 4 Temperature-time History of Specimen and Sink

VI. RESULTS AND DISCUSSION

This investigation was primarily to determine the feasibility of measuring thermal contact conductance by a method employing negligible internal thermal resistance of one of the materials providing the interface. The results are shown in Tables 1 and 2 and are plotted in Figures 5 and 6 for the cases when 10 and 100 mesh stainless steel wire screens are placed at the interface.

With the 10 mesh stainless steel wire screen at the interface, the results show that the thermal conductance, h_t , obtained by the transient method ranged from 96.04 % to 98.50 % of the thermal conductance, h_s , obtained by steady state method. From Table 1, it is seen that the transient thermal conductance approximates the steady state thermal conductance closer at low pressures than at high pressures with the 10 mesh screen at the interface. When the 100 mesh stainless steel wire screen was placed at the interface, the transient thermal conductance ranged from 79.28 % to 91.13 % of the steady state conductance. Table 2 also shows that there was closer agreement between the transient and the steady state thermal conductance at low pressures than at high pressures. In general, the values of thermal conductance obtained by the transient method are lower than the corresponding steady state values for both the 10 and the 100 mesh screen. This trend is

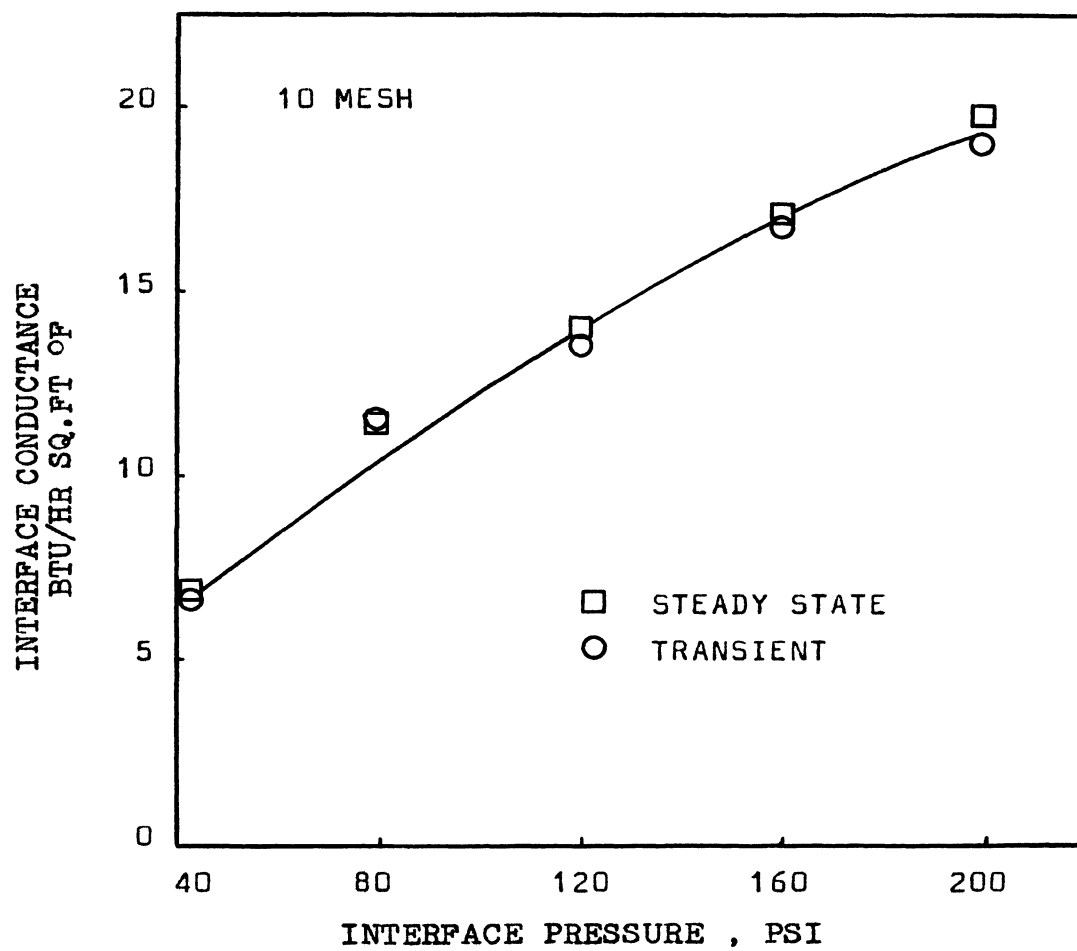


Fig. 5 Interface Conductance of Aluminum Junction
Sandwiched with Stainless Steel Wire Screen

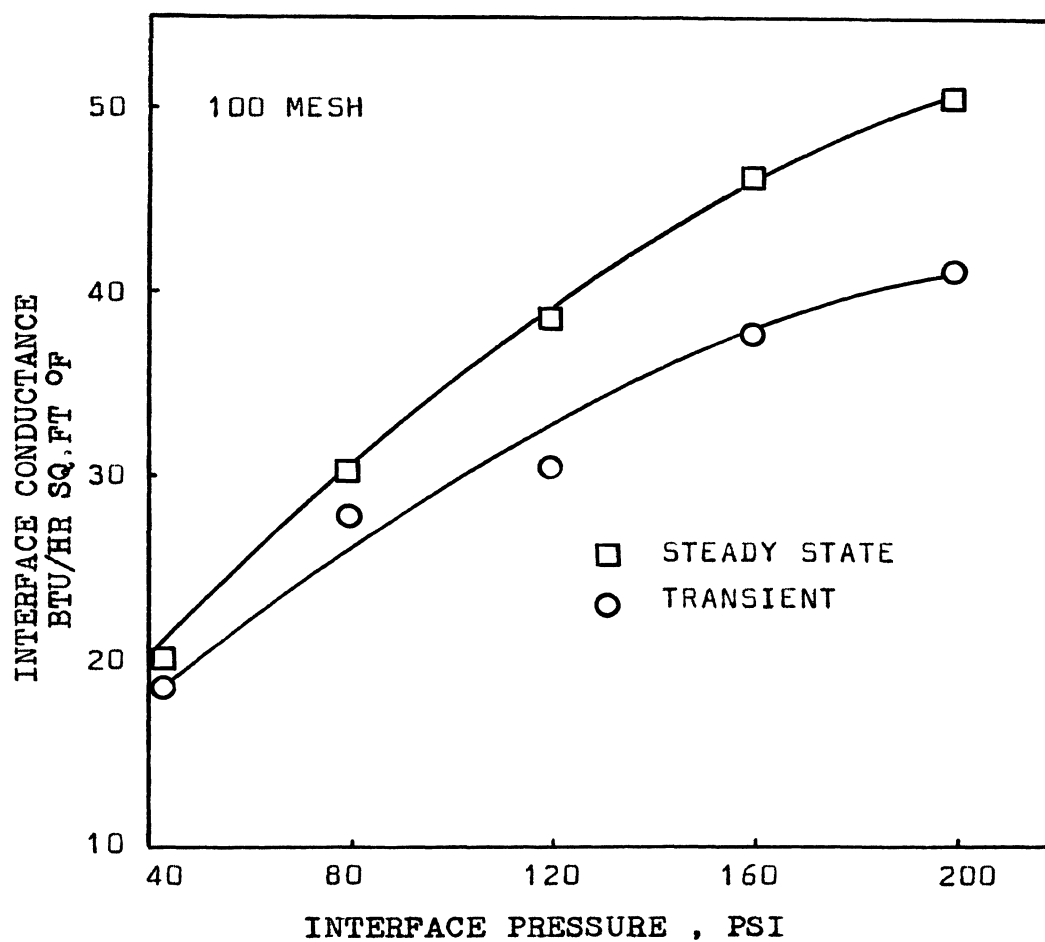


Fig. 6 Interface Conductance of Aluminum Junction
Sandwiched with Stainless Steel Wire Screen

TABLE 1
10 MESH SCREEN

PRESSURE	h_s	h_t	h_t/h_s	BIOT NUMBER
40	6.85	6.71	0.9810	0.00079
80	11.39	11.21	0.9850	0.00132
120	13.99	13.52	0.9663	0.00162
160	17.15	16.75	0.9768	0.00199
200	19.74	18.96	0.9604	0.00228

TABLE 2
100 MESH SCREEN

PRESSURE	h_s	h_t	h_t/h_s	BIOT NUMBER
40	20.27	18.07	0.9000	0.00235
80	30.39	27.70	0.9113	0.00352
120	38.56	30.56	0.7928	0.00446
160	46.21	38.50	0.8333	0.00535
200	50.65	41.15	0.8124	0.00586

attributed to the fact that in the transient method, the interface mean temperature decreases with time. Hence, since the absolute interface conductance decreases with the decrease of the mean interface temperature, the average thermal conductance as obtained by the transient method is lower than the steady state conductance which corresponds to the initial mean interface temperature.

The interface conductances were higher when the 100 mesh screen was at the interface than when the 10 mesh screen was at the interface. This behavior is due to the fact that the actual metallic contact area is greater for the 100 mesh screen than for the 10 mesh screen. Since the contact points at the interface occur only where the wire screen weave overlaps, the contact area increases with the mesh size.

In Tables 1 and 2, the Biot number which is based on the steady state interface conductance, h_s , is shown. The Biot number is calculated from the relation

$$\text{Biot number} = h_s L / k$$

where h_s is the steady state thermal conductance, L is the significant length obtained by dividing the volume of the specimen by the area of the specimen exposed to the heat transfer, and k is the thermal conductivity of the specimen. The Biot number is a measure of the relative impor-

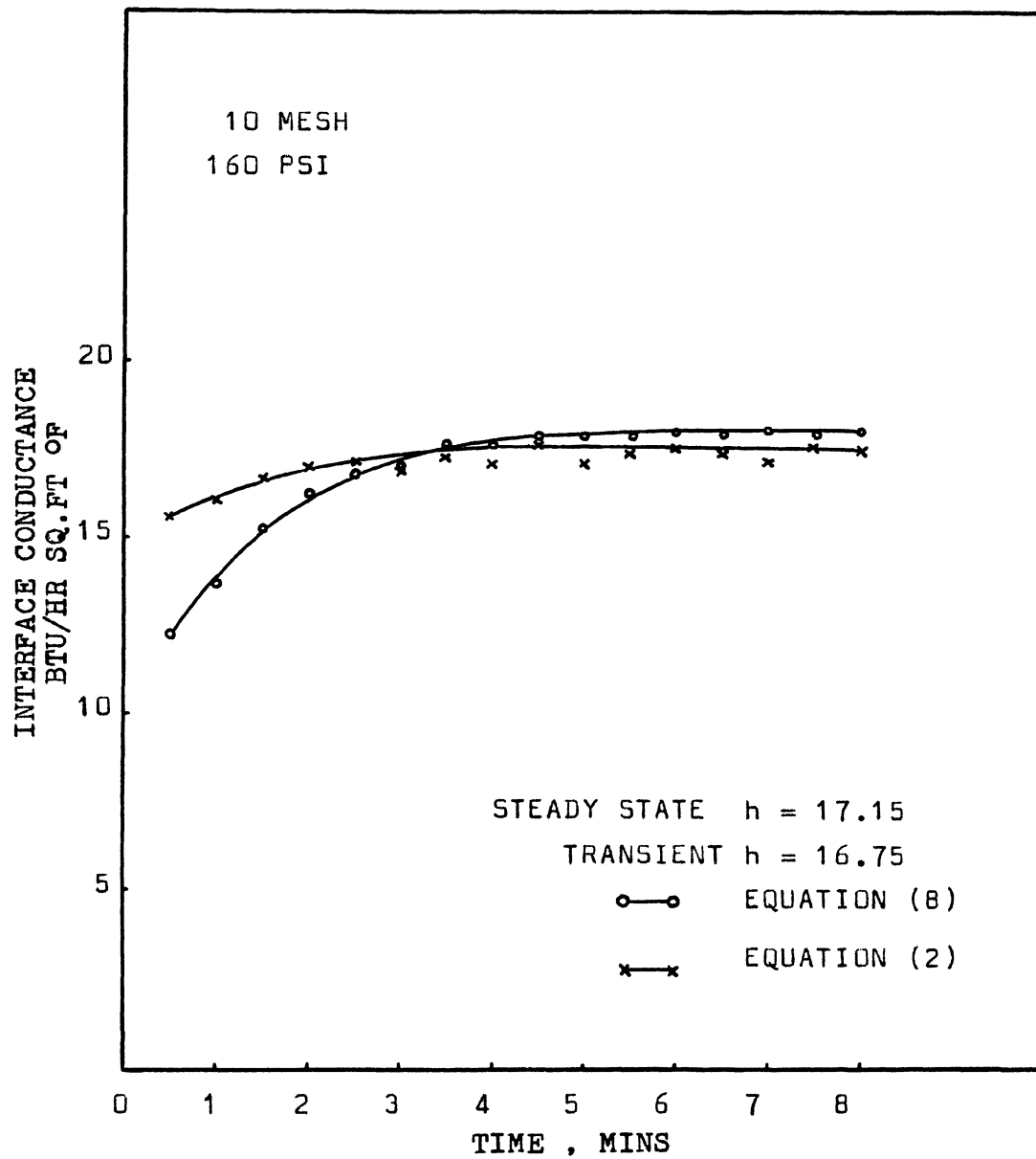


Fig. 7 Interface Conductance of Aluminum Junction
Sandwiched with Stainless Steel Wire Screen

tance of the thermal resistance within a solid body and represents the ratio of the internal to the external thermal resistance. The results show that the transient conductance agrees very closely with the steady state conductance for Biot numbers less than 0.0020. For Biot numbers above 0.0020, the transient conductance begins to deviate rapidly from the steady state conductance as the Biot number increases.

The transient conductance plotted in Figures 5 and 6 represents the average conductance at the interface. Figure 7 shows a variation of conductance with time as obtained from equations (2) and (8) respectively with the 10 mesh-160 psi run. The plots show that the conductances as given by equations (2) and (8) approach asymptotic values which are within 3 % of each other. The asymptotic values of conductance agree very well with the steady state and the transient values of conductance reported for the 10 mesh-160 psi run. The plots in Figure 7 indicate that equation (8) yields conductances which compare much better with steady state values if the first part of the curve is neglected. It seems reasonable to neglect the first part of the curve because the errors introduced by the evaluation of the slopes and by the time lag of the thermocouple are much more pronounced in this region of the curve.

Figure 8 shows a plot of dimensionless parameters which were empirically derived by Gyorgy (8). The plot of

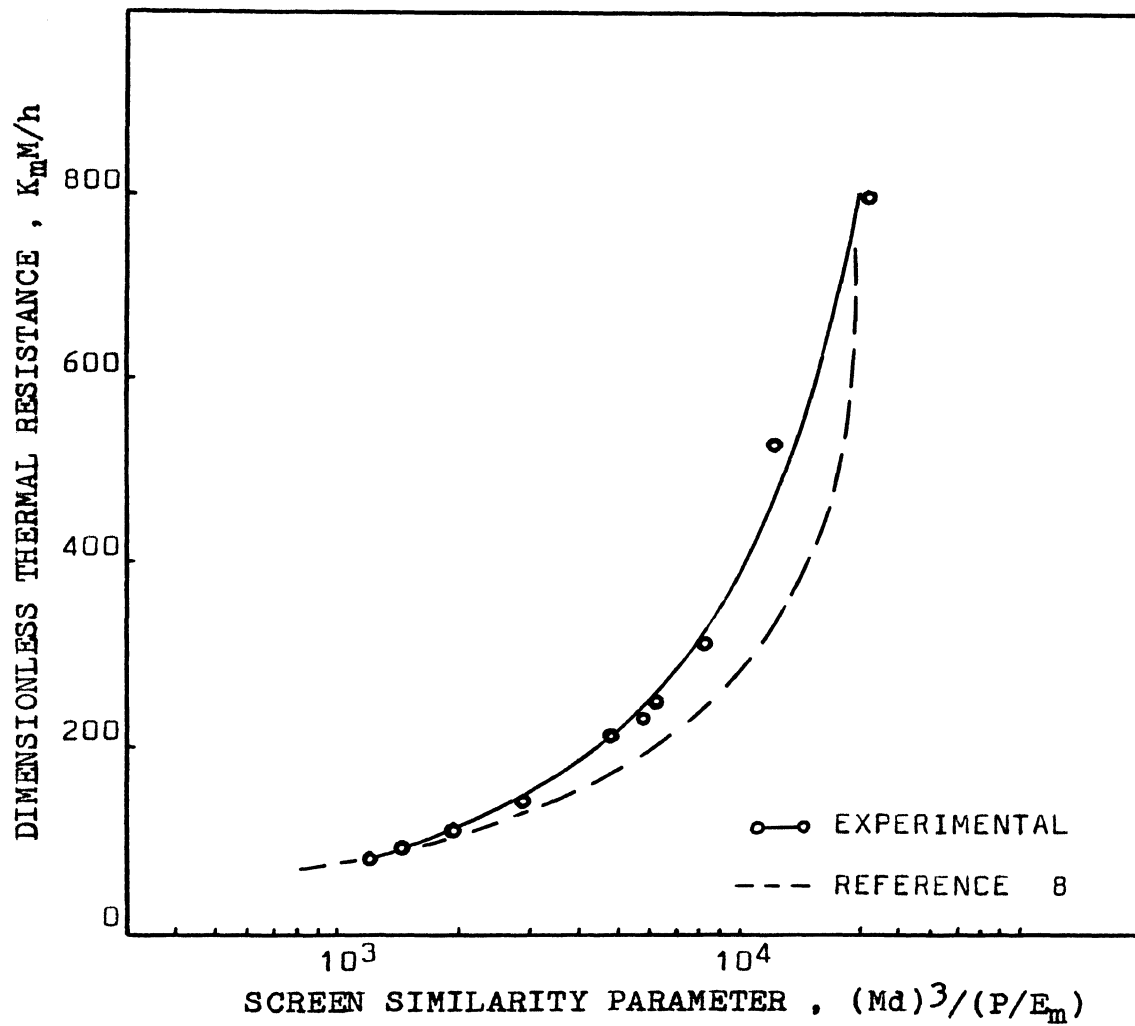


Fig. 8 Wire Screen Thermal Resistance Correlation

experimental data compares quite well with Reference 8. The agreement with Reference 8 was best at high pressures but at low pressures, the experimental results showed maximum deviation of about 25 %. This behavior is not too surprising because of extremely large variations of published data at very low pressures.

VII. CONCLUSION

The results of this investigation show that for low interface conductances between aluminum-aluminum interfaces, the transient method employing the method of negligible internal resistance can be used to determine the interface conductance quite accurately. The importance of this new method cannot be overemphasized considering the many applications where it could be extremely useful. In numerous engineering applications, such as thermal isolation, thermal contact conductance is desired to be very low, usually in the range between 0.0 and 10.0 Btu/hrsqt⁰F. In such instances, this method will provide a fast and reliable estimate of the thermal contact conductance.

VIII. RECOMMENDATIONS

More sophisticated apparatus is recommended in order to improve the results. It is suggested that a loading device which is capable of applying pressures of up to 600 psi and which can be controlled from outside the vacuum chamber be used to insure that all the test runs are made under the same vacuum environment. A method of heating the specimen which does not entail drilling a hole in the specimen and which is capable of high temperature ranges is also recommended. Further experimentation with low conductance interstitial materials such as carbon paper, mica and asbestos board should be conducted.

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X. VITA

Richard Chuka Oboka was born on January 1, 1941 in Neni-Awka, Nigeria.

He attended Secondary School and Higher School at Afikpo, Nigeria from September, 1955 to November, 1962. He attended the University of Illinois, Urbana, Illinois from where he received a B.S. in Aeronautical and Astronautical Engineering in 1967 and a B.S. in Mechanical Engineering in 1968.

He enrolled in the Graduate School of the University of Missouri at Rolla in September, 1968.

APPENDIX A1

DETERMINATION OF THE VOLUMETRIC HEAT CAPACITY AND
RADIATION HEAT LOSSES

Because the heater was installed inside the specimen in the experiment, both the specimen and the heater cooled as a unit during the transient cooling. It was thus necessary to know the thermal properties of the "specimen-heater" unit. Since heat capacity was an important parameter in the calculations, it was necessary to determine it experimentally for the specimen-heater unit. It was also necessary to estimate the radiation losses to see if the neglect of radiation in the governing equations was justifiable.

In a vacuum environment, the energy equations for the specimen-heater unit are:

$$cV(dT / dt)_h + Q_r = Q_{sup} \quad (1)$$

for the case when the unit is being heated and

$$cV(dT / dt)_c + Q_r = 0 \quad (2)$$

for the case when the unit is cooling. In the equations, Q_{sup} is a constant rate of heat supplied during heating, Q_r is the rate of radiation heat loss, $(dT / dt)_h$ and $(dT / dt)_c$ are the rates of temperature change during heating and cooling respectively. Solving equations (1)

and (2) yield

$$c = \frac{Q_{\text{sup}}}{V((dT/dt)_h - (dT/dt)_c)} \quad (3)$$

and

$$Q_r = \frac{-Q_{\text{sup}}(dT/dt)_c}{((dT/dt)_h - (dT/dt)_c)} \quad (4)$$

where $(dT/dt)_h$ and $(dT/dt)_c$ are calculated at the same temperature or range of temperatures.

An experiment was carried out to determine Q_r and c as follows: The specimen-heater unit was isolated or suspended in the vacuum chamber by means of the electrical lead wires to the heater and the thermocouple wires on the specimen in an environment similar to that which prevailed throughout the rest of the experiment. A constant power supply was then used to heat the unit and its temperature-time history was recorded with a chart recorder. After the temperature reached some arbitrary value, the power supply was turned off and the specimen-heater unit was allowed to cool by means of radiation only while its temperature-time was recorded.

From the temperature-time histories of the unit during heating and cooling, $(dT/dt)_h$ and $(dT/dt)_c$ were calculated and used in equations (3) and (4) to evaluate the volumetric heat capacity, c , and the radiation heat loss, Q_r .

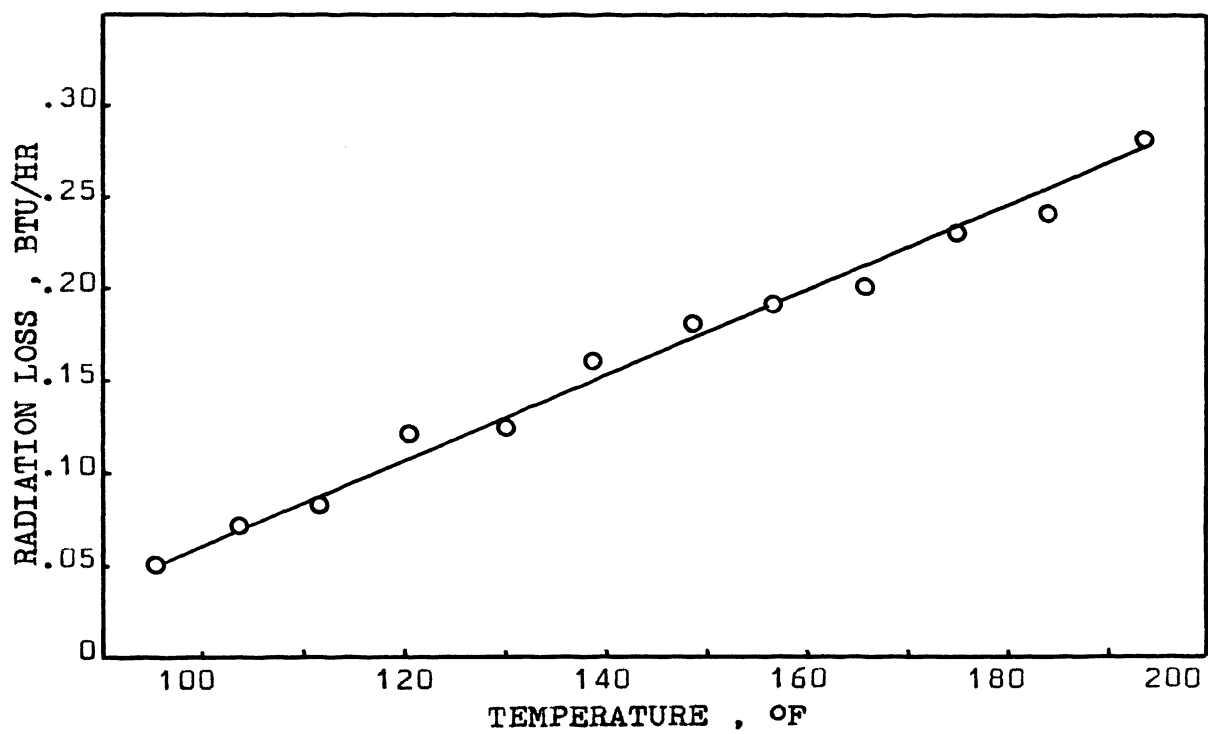


Fig. 9 Radiation Loss Vs. Temperature of Specimen

The volumetric heat capacity, c , was found to be 70.54 Btu / cu. ft $^{\circ}\text{F}$. The radiation heat loss, Q_r , is shown in Figure 9 as a function of temperature. It was found that the maximum radiation heat loss during the entire experiment represented less than 2.0 % of the heat flowing through the interfaces. The assumption of negligible radiation heat loss was thus justifiable.

APPENDIX A2

CALCULATION OF CONDUCTANCE

The temperature of the specimen at any instant during cooling is expressed analytically as

$$\begin{aligned}
 T(t) = & (T_0 - (2C_1 / m^2 - C_2 / m + C_3))e^{-mt} \\
 & + C_1 t^2 + (C_2 - 2C_1 / m)t + 2C_1 / m^2 \\
 & - C_2 / m + C_3
 \end{aligned} \tag{1}$$

The temperature-time history of the specimen was obtained experimentally; hence, at any instant of time, the only unknown quantity in equation (1) is m , where

$$m = hA / cV \tag{2}$$

A value of m is sought so that equation (1) gives the best fit to the experimental data.

Let T_i be the experimental value of the temperature of the specimen at time t_i , $i = 0, 1, 2, \dots, n$, where $t_i - t_{i-1}$ is constant. Let

$$I_1 = \int_{t_0}^{t_n} T_i dt \tag{3}$$

and

$$I_2 = \int_{t_0}^{t_n} T(t) dt \tag{4}$$

where $T(t)$ is given in equation (1). Then for best fit, it is required that

$$I_1 - I_2 = 0 \quad (5)$$

In order to solve for m , the following iterative scheme is employed:

- (i) Using the experimental data obtain I_1 from equation (3).
- (ii) Assume a value for m .
- (iii) Obtain I_2 from equation (4).
- (iv) Evaluate

$$e = \frac{\text{ABS} (I_1 - I_2)}{I_1} .$$

- (v) If $e > 10^{-4}$, assume another value of m and repeat steps (iii), (iv), and (v). If $e \leq 10^{-4}$, then m is a solution. The thermal contact conductance is then calculated using equation (2).

APPENDIX A3

UNCERTAINTY ANALYSIS

The uncertainty is the estimate of experimental error in measured quantities. The uncertainty in the measured values leads to the uncertainty in the calculated values. The uncertainty of some pertinent parameters involved in this investigation are discussed below.

The steady state conductance, h_s , and the transient conductance, h_t were calculated from the equations

$$h_s = Q/(A\Delta T) \quad (1)$$

and

$$h_t = mcV/A \quad (2)$$

The uncertainty in the conductances are given by

$$(dh_s/h_s)^2 = (dQ/Q)^2 + (dA/A)^2 + (d\Delta T/\Delta T)^2 \quad (3)$$

and

$$(dh_t/h_t)^2 = (dm/m)^2 + (dc/c)^2 + (dV/V)^2 + (dA/A)^2 \quad (4)$$

The heat supplied, Q , was calculated from the voltage and the current recorded during each test run. The smallest scale divisions of the meters were 2 milliamperes and 5 volts. The estimated error in temperature measurements was about 1 °F. The resulting uncertainty in the conductances ranged from 3.0 to 8.5 %.

The interface pressures were calculated from the equation

$$P = (WL_1)/(AL) \quad (5)$$

for which the uncertainty expression is given by

$$(dP/P)^2 = (dW/W)^2 + (dA/A)^2 + (dL_1/L_1)^2 + (dL/L)^2 \quad (6)$$

The error in measuring the load, W , was estimated to be about 1.0 lb. and the error in measuring the moment arms, L_1 and L_2 , of the loading system and the diameter of the specimen was about 0.001 inch. The resulting uncertainty in the interface pressure ranged from 5.0 to 12.0 %.